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Design of an Acoustic Muffler Prototype for an Air Filtration System Inlet on International Space Station

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1. INTRODUCTION

The Zarya control module, also known by the technical term Functional Cargo Block (FGB), was the first International Space Station (ISS) element placed in space and provided the "foundation" for assembly of the other ISS elements. The FGB provides systems for propulsion; guidance, navigation and control; electrical power; communications; partial life support functions; and thermal control. The Russian-provided crew living quarters and early station core known as the Service Module enhanced or replaced many functions of the Zarya. The FGB is now mainly used for its storage capacity and external fuel tanks. The FGB features an air filtration system of which the two inlets, one on each side of the cabin, have filters that remove dust-type particles from the module's atmosphere. Behind each inlet are two encased, two-bladed fans, which circulate the air. Noise generated by the air filtration system exceeded the applicable acoustic continuous noise criteria in the octave bands from 250 Hz to 8000 Hz. The design, construction and testing of a prototype muffler was initiated to provide a viable noise abatement mechanism.

2. AIR FILTRATION SYSTEM

The two encased fans behind each of the FGB air filtration system inlets feature two 170-mm diameter blades. Each fan was installed in a 194-mm diameter cylindrical case and mounted on top of a rectangular box containing the dust filter (Figure 1). The box dimensions were approximately 550 mm for the length, 230 mm for the width and 88.6 mm for the height. The fan blades were located at a distance of 319 mm from the inlet grating. A distance of 259 mm separated the axes of the two fans. The inlet inflow area covered 18644 mm^2. The blade passage frequency was calculated to be 100 Hz at 3000 rotations per minute (rpm). The fans were mounted behind a 50-mm diameter hub, which was attached to the case housing by a 40-mm thick strut as shown in Figure 2.

3. PROBLEM ASSESSMENT

Only limited experimental acoustic data were available. Measurements in the FGB at 1-meter distance from the air filtration system inlets indicated that sound pressure levels exceeded the applicable acoustic requirements for continuous noise (Figure 3). The differences between measurements and requirements are also listed in Table 1 showing an exceedance of up to 9.2 dB over the octave band frequency range from 250 Hz to 8000 Hz. A narrowband spectrum was available (Figure 4) for one of the fans in the same air filtration system showing a major tonal component at 360 Hz. Figure 4 also shows some peaks at other discrete frequencies and broadband noise at lower amplitudes. The origin of the major tonal component at 360 Hz was unknown, but was not related to the rotational (50 Hz) or the blade passage frequency (100 Hz) at 3000 RPM. The strut on which the hub is mounted significantly disturbs the inflow to the fan blades. When the blades move through the wake behind the strut an impulsive noise could be generated related to the blade passage frequency. The disturbed inflow to the fan is also a broadband noise source as the increased turbulence interacts with the pressure fluctuations on the fan blades. A fan designed to operate behind aerodynamic vanes or in front of a hub support structure would be a more advantageous design from an acoustic point of view. However, changes in the design of the fan were not an option. In addition to the aerodynamic sources of noise, structureborne noises from the motor, from rotor unbalance, from misaligned bearings or from other mechanical sources could be generated and radiated along a structural path. Since not enough information was available, a muffler was designed to attenuate the 360 Hz tone and to reduce the broadband noise content in an attempt to achieve compliance with the applicable continuous noise requirements.

4. MUFFLER DESIGN

The muffler for the air filtration duct inlet was subject to geometric design constraints. The total thickness of the muffler was limited to 80 mm and a space of at least 20 mm was required between the filter and the inlet surface. To be effective for both tonal and broadband noise sources the muffler was designed to serve several noise abatement functions, including noise transmission loss, structural damping, sound absorption, tone attenuation, and flow guidance to ensure a smooth path for the moving air. Because of schedule constraints, design and fabrication complexity, and the need to develop a quick, inexpensive way to evaluate a proof-of-concept design, the muffler was constructed by a Stereolithography Apparatus (SLA) process, which produces very accurate form/fit prototypes with a quality surface finish. The SLA process uses an ultraviolet (UV) laser system to selectively cure an epoxy photopolymer into a solid object. The muffler was composed of 3.175-mm thick SLA material and covered with 3.175-mm thick aluminum material. Figure 5 presents a top view of the SLA model and a bottom view of the SLA prototype inside the aluminum cover.

A. Noise Transmission Loss and Diffraction

Noise attenuation was achieved by placing the muffler as an obstruction between the fan noise sources inside the air filtration system and the receiving space in the FGB cabin. The straight line of sight noise path between the fans and the cabin interior space was interrupted. Angled air inlets were designed to direct the flow around two 150 mm by 180 mm rectangular, 3.175-mm thick, aluminum panels. The flow inlets around the two aluminum panels are shown in the computer aided design (CAD) drawings of Figure 6. The mass-law field noise transmission loss of the aluminum panel increases at 6 dB per octave, from 7.3 dB at 63 Hz to 49.3 dB at 8000 Hz. Transmitted noise will thus be of low acoustic intensity. Most of the noise reaching the FGB cabin will bend around the aluminum center panels, along a path through the flow inlets, where diffraction is the main noise attenuation mechanism.

B. Structural Damping

A 150 mm by 180 mm rectangular layer of visco-elastic material was applied to the inside of the two rectangular aluminum panels to dampen structural vibrations and to minimize acoustic radiation. Visco-elastic tape could also be applied to other parts of the muffler, as it is effective in damping structural borne sound without adding appreciable weight to the structure.

C. Acoustic Absorption

Open-cell melamine acoustic foam was applied to the visco-elastic layer on the inside of the aluminum rectangular panels as indicated in Figure 6. Due to the height constraints of the muffler only a 25.4-mm thick layer could be installed. The ability of 25.4-mm thick foam to absorb sound decreases rather rapidly below 1000 Hz, as indicated by sound absorption coefficients of 0.73 at 500 Hz, 0.29 at 250 Hz and 0.08 at 125 Hz.

The foam carries only a very small weight penalty. The foam has excellent materials outgassing and flammability characteristics.

D. Tone Attenuation

Helmholtz resonators were incorporated in the design of the inlet to attenuate the 360 Hz tonal peak in the frequency spectrum. Figure 6 shows the Helmholtz resonator cavities in the SLA material. An aluminum cover with rectangular cutouts for the flow inlets will close the Helmholtz cavities. Figure 7 depicts the aluminum cover mounted onto the bottom of the muffler prototype. The Helmholtz resonance for a side branch is given in Reference 1 by:

$$f_{Helmholtz}=c/(2\pi)\ \sqrt{[A/(LV)]}$$

where c is the speed of sound [m/s], A is the area [m^2] of the resonator inlet vent, L is the effective length [m] of the inlet and V is the volume [m^3] of the resonator cavity. A bushing with a radius of 6.35 mm was installed to be the Helmholtz resonator inlet vent. The resonator was tuned to the 360 Hz peak by selecting an inlet length of 6.35-mm and a resonator volume of 170.9 mm^3. Eleven Helmholtz resonators, all tuned to the same 360 Hz resonance, were installed around each of the two flow inlets (Figure 6).

The effective length, used in the calculation of the Helmholtz resonance frequency, is based on the fact that not only the column of air precisely fitting the inlet is resonating, but that also some of the air on either side of that air column will be part of the resonance. The effective length of the resonating column of air depends on the diameter and length of the resonator inlet, and the flow conditions on either side of the inlet. The actual Helmholtz resonance frequency might thus be somewhat different than the calculated one. Changes in temperature also will affect the resonance frequency. The Helmholtz resonance was calculated for a temperature of 20^o Celsius. A change in temperature of 3 degrees will vary the resonance frequency by as much as 2 Hz, due to a change in the speed of sound. Only after installation of the muffler onto the air handling system inlet will it be possible to measure the exact Helmholtz resonance frequency and the effectiveness of the resonators. To enable adjustments in the Helmholtz resonance frequencies after installation of the muffler the SLA material was designed with holes that accept bushings with fixed inner (12.7 mm) and outer diameter but of varying length. By choosing a length of 7.98 mm for the bushing (inlet vent) the resonance would occur at 345 Hz. A length of 5.01 mm (still longer than the thickness of the SLA material) would correspond to a resonance of 375 Hz. The parameters for the three cases discussed here are listed in Table 2. This approach facilitates tuning the resonators to the desired frequency even after installation by simply fitting the appropriate length bushings. It would also be possible to tune to more than one frequency although the effectiveness of the resonator system would be less as fewer resonators are used in attenuating each frequency.

E. Flow Guidance

A fairing was designed to ease the flow through the muffler avoiding sharp discontinuities and abrupt flow path changes. Minimizing the turbulence of the air that flows into the fans diminishes the aerodynamic noise generation. The total muffler inlet area was 18644 mm^2 and the cross-sectional area along the flow path through the muffler was kept constant.

5. MEASUREMENTS

The muffler was tested in two different configurations. . The purpose of the first test configuration was to measure the broadband noise insertion loss of the muffler without the Helmholtz resonators. In the second test the noise attenuation due to the Helmholtz resonators, when exposed to a 360 Hz tone, was measured.

A. Broadband attenuation

The muffler was affixed to a 19-mm thick piece of plywood with a cutout to simulate the arrangement of the air filtration system inlet. The plywood/muffler combination was mounted over a 0.914 m by 1.219 m test window in a 0.305-m thick wall, separating a small reverberation chamber from a semi-anechoic room. The plywood/muffler combination contained a wire mesh grille and air filter material to approximate the features of the actual hardware. The Helmholtz resonator vents were closed with tape. White noise was fed into an amplifier and played back by a loudspeaker in the reverberation chamber. One-third octave band data were taken at 0.6 meter from the muffler front surface. Sound pressure level measurements were conducted with and without the muffler installed. The muffler octave band insertion loss was calculated and the results are graphed in Figure 8. It is shown that between 3 dB and 8 dB insertion loss was achieved over the entire range of octave bands between 63 Hz and 8000 Hz. Direct comparison of the measured attenuation with the reduction needed to comply with the requirements (Table 1) is not relevant due to the differences in noise excitation (diffuse white noise versus fan noise), noise source environment (reverberation chamber versus air filtration system duct) and the absence of a flow mechanism. In-situ measurements will be needed to substantiate compliance with the requirements.

B. Tone Attenuation

The reverberant chamber was a suitable environment for the white noise source in the broadband insertion loss measurements. In the tone attenuation measurements, however, the loudspeaker source was placed in an anechoic environment to avoid reflections, cancellation/amplification effects and acoustic standing waves patterns. A special test box was designed for this purpose with walls of 19-mm thick plywood. The test box was lined with 254-mm thick acoustic polyurethane open-cell foam to provide an almost anechoic environment at the 360 Hz test condition. The box is shown in Figure 9. The muffler prototype was mounted onto the front baffle of the box, over a cutout that simulated the air filtration inlet. A close-up of the prototype muffler mounted on the test box is depicted in Figure 10. The back of the plywood box contained two 165-mm diameter loudspeakers. The loudspeakers were separated by approximately the same distance (259 mm center-to-center) as the fans in the air filtration system (Figure 1). The 319 mm distance from the loudspeakers to the muffler on the box was the same as the distance between the fan and the air filtration grill in the actual configuration. The Helmholtz resonator vents were closed before the start of the acoustic tests. A tripod-mounted sound level meter was placed at a distance of 0.6 meter in front of the muffler between the two inlet openings (Figure 9). A 360 Hz sine wave was generated, played back through the loudspeakers in the box, measured by the sound level meter and analyzed by a real-time analyzer. The narrow-band measurements were performed with a 1.5 Hz bandwidth resolution. The measurements were repeated with the Helmholtz resonator vents uncovered. The delta sound pressure level of the acoustic reactive attenuation by the resonators is illustrated in Figure 10 for the left and right inlet ports. Up to 13.9 dB attenuation was obtained for the resonators around the right muffler inlet and up to 11.6 dB for the resonators around the left muffler inlet. The resonator system with the highest attenuation (right inlet) had the least effective frequency range (24 Hz at 6 dB down from its peak attenuation). The resonator system at the left muffler inlet had an effective frequency region of 37.5 Hz at 6 dB down from its peak attenuation. Applying sound absorption materials inside the Helmholtz resonators was considered to broaden the attenuation range, but that would have also resulted in lower peak attenuation. Airflow, which was not included in the measurements, will affect the Helmholtz resonance frequency and therefore the noise attenuation characteristics of the muffler. Other factors that may affect the accuracy of the measurements and the resulting attenuation include the loudspeaker generated noise source (rather than aerodynamically generated fan noise), cancellation/amplification effects by the tones originating from each of the two speakers, structureborne noise and flanking paths.

Combining the broadband insertion loss with the tone attenuation data suggests that the muffler provides significant opportunities to attenuate the noise emanating from the air filtration unit without altering the source (fans), the path or the performance of the system. The prototype muffler design was realized within the geometric constraints of the total, existing system. Further improvements could be accomplished at the noise source, by redesigning the hub, incorporating quieter fans, designing a more streamlined flow path and creating more physical space to implement noise abatement measures.

6. SUMMARY

A muffler was designed to attenuate the noise of an air filtration system on International Space Station. The concept included noise transmission loss, structural damping, acoustic absorption, tone attenuation and flow guidance. Temperature and effective length of the resonator inlet were recognized as uncertainties in the acoustic performance of the muffler. Variable length bushings were used to enable tuning of the muffler and maximize the Helmholtz resonator efficiency after installation. Broadband and narrow band noise insertion loss measurements validated the muffler as an effective noise abatement tool for the ISS air filtration system.

7. ACKNOWLEDGEMENTS

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8. REFERENCE

Fundamentals of Acoustics - 2nd Ed. Lawrence E. Kinsler & Austin R. Frey John Wiley & Sons, Inc. © 1962 (p. 187, 193)

9. TABLES

Table 1. Difference between measured sound pressure levels ($SPL_{measured}$) and the continuous noise requirements (SPL_{spec})

Octave band, Hz	$SPL_{measured} – SPL_{spec}$, dB
63	-5.9
125	-0.6
250	9.6
500	8.1
1000	5.3
2000	9.2
4000	3.2
8000	3.4

Table 2. Helmholtz resonance frequencies of fixed resonator inlet design compared with flexible bushing concept

Resonator	Fixed design	Bushing concept	Bushing concept
Volume, mm^3	171	171	171
Inlet radius, mm	6.35	6.35	6.35
Inlet length, mm	6.35	7.98	5.01
Frequency, Hz	360	345	375

10. FIGURES

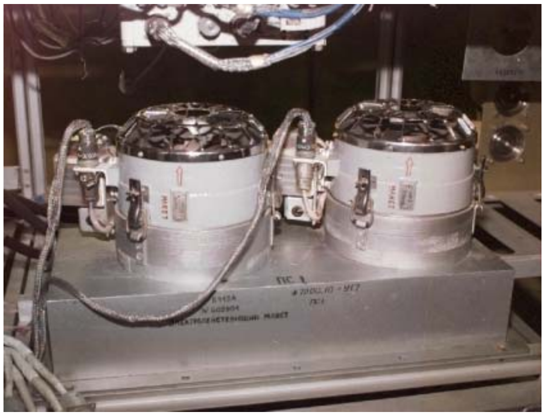

Figure 1. FGB air filtration fans

Figure 2. FGB fan hub

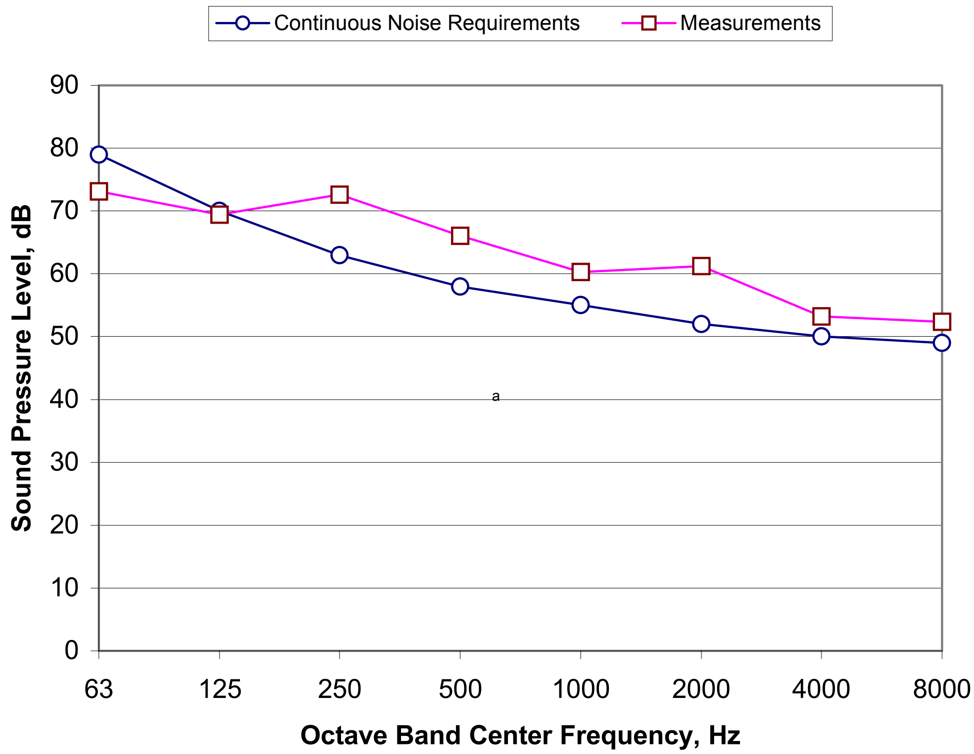

Figure 3: Measured sound pressure levels compared with continuous noise requirements

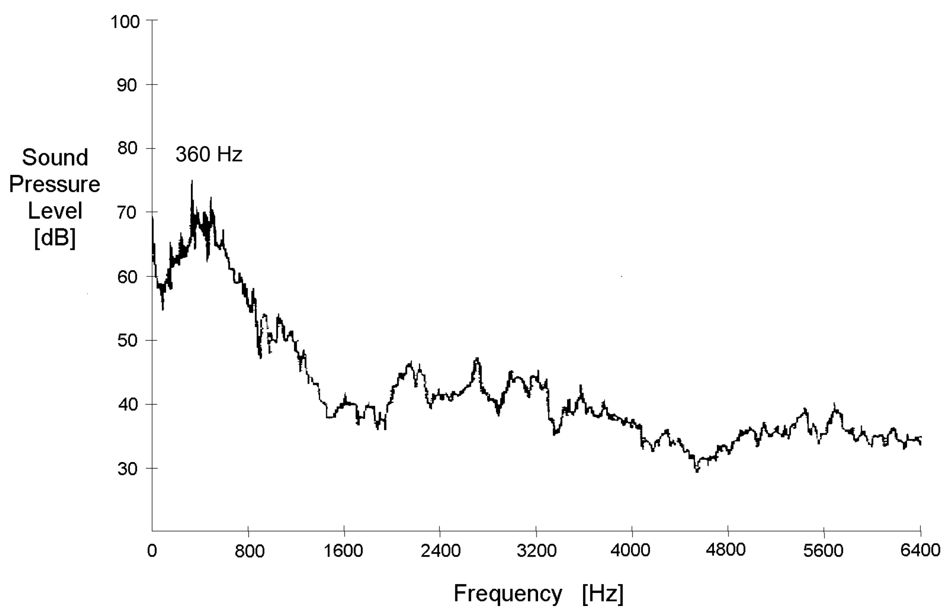

Figure 4: Narrowband spectrum of one air filtration system inlet fan

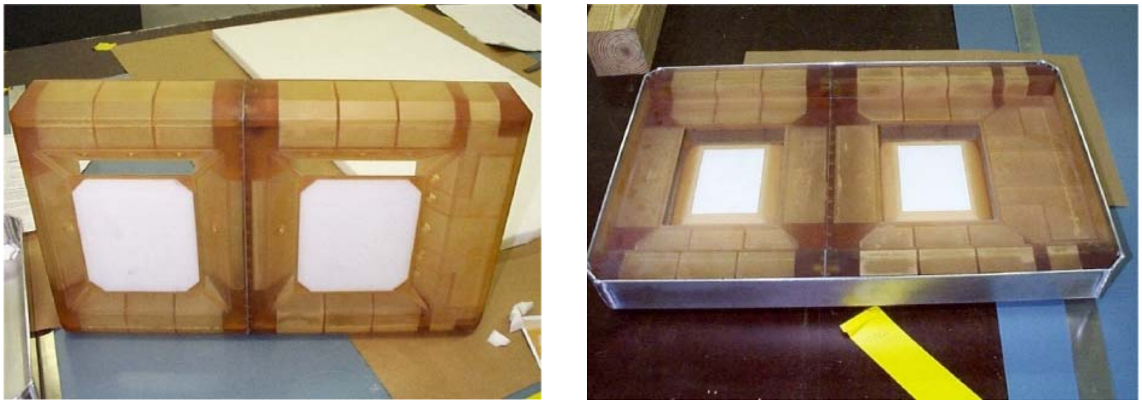

Figure 5. Muffler prototype showing views of top (SLA material only) and bottom (with aluminum cover)

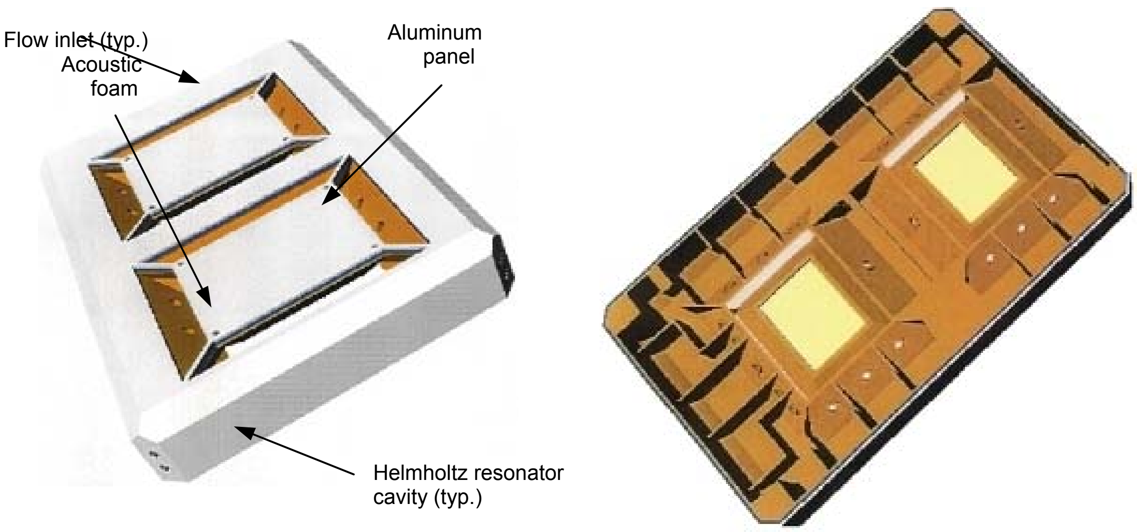

Figure 6. Muffler computer aided design drawings showing views of top and bottom

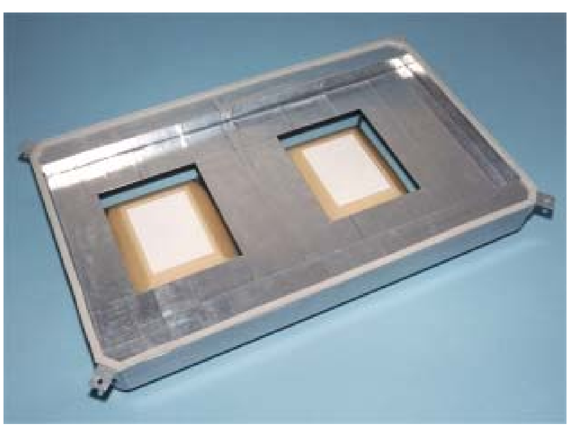

Figure 7: Muffler prototype with bottom cover

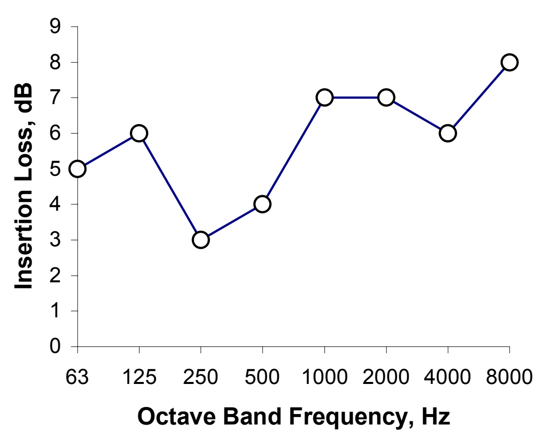

Figure 8: Muffler broadband insertion loss

Figure 9. Test box and experimental setup for Helmholtz resonator testing of the prototype muffler

Figure 10. Close-up of prototype muffler mounted on the test box

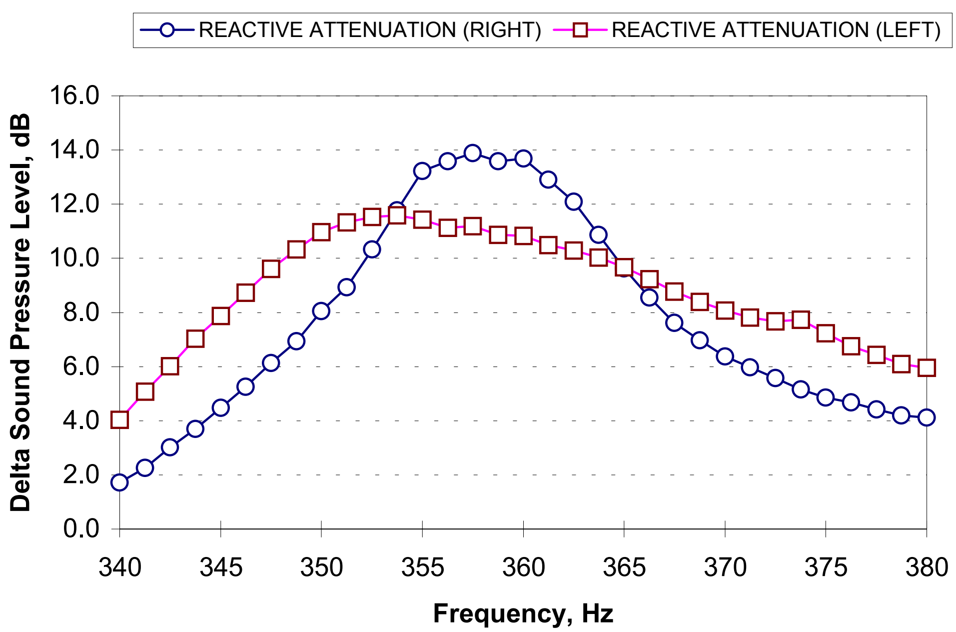

Figure 11. Helmholtz resonators acoustic attenuation